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(54) Continuously variable transmission

(57) A continuously variable transmission sets the spring loads of the springs that urge the movable sheaves of primary-side and secondary-side pulleys, as indicated by curves f,c. Thereby the pulley ratio for the vehicle stop range (the P or D range) becomes a pulley ratio (1) approximate to the pulley ratio r_N corresponding to the neutral state. This setting prevents a vehicle from taking off backward when a driver shifts from the vehicle stop range to the D range immediately after starting the engine, that is, before the minimum required fluid pressure is supplied. The setting also causes the pulley ratio (2) occurring after the fluid pressure supply to become approximate to the pulley ratio r_N. Thus the continuously variable transmission eliminates a vehicle takeoff response delay caused by a pulley ratio change.

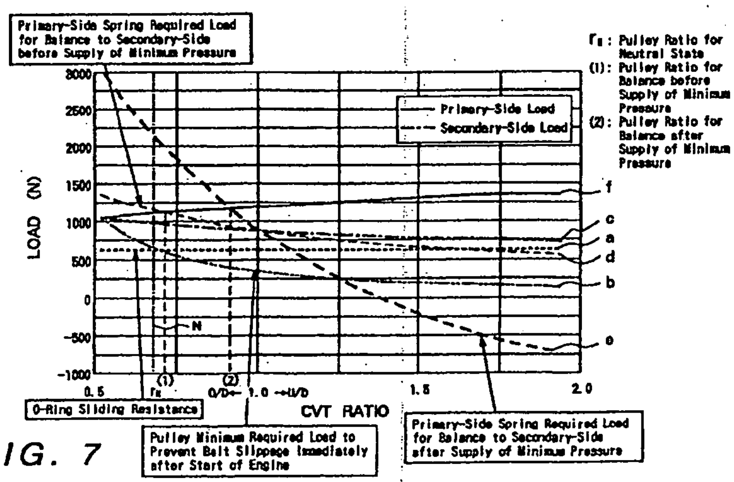

FIG. 7

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Description

The invention relates to a continuously variable transmission installed in a motor vehicle or the like and, more particularly, to a continuously variable transmission comprising a constant speed transmission apparatus and planetary gears combined.

Among automatic transmissions installed in motor vehicles or the like, continuously variable transmissions are drawing attention as a transmission other than multi-speed automatic transmissions providing three or four forward speeds.

For example, a continuously variable transmission (a first conventional art example) disclosed in U.S. Patent No. 4,467,670 employs a belt-type continuously variable transmission apparatus comprising a first pulley provided at the input side, a second pulley at the output side, and a belt wound around them, wherein the movable sheaves of the first and second pulleys are hydraulically moved in the directions of the axis to change the clamping forces on the belt, thereby suitably changing the pulley ratio. This construction enables a vehicle to perform continuously variable transmission. In this example construction, the pulley ratio shifts toward the overdrive side if the hydraulically achieved clamping force of the first pulley is increased over that of the second pulley, and it shifts toward the underdrive side if the clamping force of the second pulley is higher.
Furthermore, this construction comprises a return spring for the second pulley that adds to the second pulley clamping force to actively shift the pulley ratio toward the underdrive side, thereby preventing a vehicle re-takeoff response delay based on the belt shift. In ordinary constructions, one of the first and second pulleys is provided with a return spring to add to the belt clamping force, thus preventing the belt from slipping even when the hydraulically achieved clamping force is zero and preventing the belt from falling off the pulleys.

A continuously variable transmission (a second conventional art example) disclosed in U.S. Patent No. 4,644,820 comprises, between the input and output shafts, a belt-type continuously variable transmission apparatus, a chain-drive constant speed transmission apparatus, and a planetary gear. By suitably combining the operations of the devices and the gear set, the continuously variable transmission enables torque circulation, changing over the forward drive state, the neutral state, and the reverse drive state of the vehicle corresponding to the continuous variation of the pulley ratio.

However, if the second pulley is provided with a return spring to add to the clamping force as in the first example, the pulley ratio converges in the underdrive side (the maximum pulley ratio side). Conversely, if the first pulley is provided with a return spring, the pulley ratio converges in the overdrive side (the minimum pulley ratio side).

In such a construction of the second example, it is necessary to establish the neutral state to take off the vehicle, that is, it is necessary to set a predetermined pulley ratio corresponding to the neutral state. Since the predetermined pulley ratio is between the maximum pulley ratio and the minimum pulley ratio, the pulleys must be operated to change the pulley ratio from the maximum or minimum ratio to the predetermined ratio during takeoff operation if only one of the pulleys is provided with a return spring. Therefore, there is a danger that a takeoff response delay may be caused by a shift of the pulleys.

Accordingly, it is an object of the invention to provide a continuously variable transmission that changes over the forward drive state, the neutral state and the reverse drive state of a vehicle corresponding to changes in the pulley ratio and that prevents a takeoff response delay based on a shift of a pulley and thus enables a smooth takeoff.

According to the invention, there is provided a continuously variable transmission comprising: an input shaft drivingly connected to an engine output shaft; an output shaft drivingly connected to a wheel; a belt-type continuously variable transmission apparatus having a first pulley drivingly connected to the input shaft, a second pulley provided on the side of the output shaft, a belt wound around the two pulleys, and axial force applying means for applying axial force to the pulleys to change the pulley ratio; and a planetary gear having a first rotary element connected to the input shaft, a second rotary element connected to the second pulley, and a third rotary element connected to the output shaft. The pulley ratio is continuously changed to correspondingly achieve a forward drive state, a neutral state, and a reverse drive state of a vehicle through the output shaft. The axial force applying means comprises first urging means for constantly urging the first pulley to apply an axial force thereto, and second urging means for constantly urging the second pulley to apply an axial force thereto. The urging forces of the first and second urging means are set such that a pulley ratio r_1 becomes approximate to a pulley ratio r_N, where the pulley ratio r_1 is a pulley ratio achieved by the axial force applying means in a vehicle stop range in which no torque is transmitted to the output shaft, and the pulley ratio r_N is a pulley ratio that achieves the neutral state.

Preferably, each of the first and second pulleys comprises a stationary sheave and a movable sheave that clamp the belt therebetween, and the first urging means and the second urging means are a first spring and a second spring, respectively, that force the corresponding movable sheaves to clamp the belt.

It is also preferred that the axial force applying means comprise a first hydraulic chamber that receives a fluid pressure supply to urge the corresponding movable sheave in cooperation with the first spring, and a second hydraulic chamber that receives a fluid pressure supply to urge the corresponding movable sheave in cooperation with the second spring.

It is preferred that the urging forces of the first and second springs be set such that the pulley ratio r_1 becomes

approximate to the pulley ratio r_N before each of the first and second hydraulic chambers is supplied with a minimum required fluid pressure.

Preferably, the pulley ratio r_1 is a pulley ratio that achieves the forward drive state.

Preferably, the continuously variable transmission of the invention further comprises a running clutch that is engaged in a vehicle driving range. The urging forces of the first and second springs are set such that within a period between the start and the completion of engagement of the running clutch, the pulley ratio occurring after each of the first and second hydraulic chamber is supplied with the fluid pressure converges to the pulley ratio r_N.

It is also preferred that the urging forces of the first and second springs be set such that the pulley ratio r_1 becomes approximate to the pulley ratio r_N after each of the first and second hydraulic chambers is supplied with a minimum required fluid pressure.

The foregoing and further objects, features and advantages of the invention will become apparent from the following description of preferred embodiments with reference to the accompanying drawings, wherein:

Fig. 1 is a front sectional view of a continuously variable transmission according to the invention;
Fig. 2 is a side view of the continuously variable transmission;
Fig. 3 is a speed graph of the continuously variable transmission;
Fig. 4 shows the engagement states of the clutches;
Fig. 5 is a graph indicating the change of the output torque with respect to the torque ratio of the belt-type continuously variable transmission apparatus (CVT);
Fig. 6 is a graph indicating the change of the output rotational speed with respect to the torque ratio of the CVT;
Fig. 7 illustrates the spring load setting to achieve predetermined pulley ratios; and
Fig. 8 illustrates another manner of the spring load setting to achieve predetermined pulley ratios.

Preferred embodiments of the invention will be described hereinafter with reference to the accompanying drawings.

First, the overall construction of a vehicular continuously variable automatic transmission 1 according to an embodiment of the continuously variable transmission of the invention will be described with reference to Figs. 1 and 2. Fig. 1 is a combined drawing made up of a skeleton diagram and a sectional view taken along various shafts described later. Fig. 2 is a side view showing the positional relation among the shafts of the vehicular continuously variable automatic transmission 1.

The vehicular continuously variable automatic transmission (hereinafter referred to simply as "continuously variable transmission") 1 shown in Figs. 1, 2, comprises as main components: a first shaft (input shaft) 3 drivingly connected to an engine crank shaft (engine output shaft) 2, a second shaft (output shaft) 4 drivingly connected to wheels, a belt-type continuously variable transmission apparatus 11, and a planetary gear 19. The belt-type continuously variable transmission apparatus (hereinafter, referred to as "CVT") 11 comprises a primary pulley (first pulley) 7 disposed on the first shaft 3, and a secondary pulley (second pulley) 9 disposed on the second shaft 4, a belt 10 connected between the pulleys 7, 9, and axial force applying means 7c, 9c for applying axial force to the pulleys 7, 9. The planetary gear 19 comprises a carrier 19c, a sun gear 19s, a ring gear 19r as the first, second and third rotary elements.

The first shaft 3 and the second shaft 4 are disposed parallel to each other. Also disposed parallel to these shafts are a third shaft 5 formed of a countershaft, a fourth shaft 6 that forms an axle connected to front wheels. The continuously variable transmission 1 is constructed so that the rotation generated by the engine is transmitted to the front wheels by the first shaft 3, the second shaft 4, the third shaft 5, the fourth shaft, the belt-type continuously variable transmission apparatus 11, the planetary gear 19, etc., and it is continuously variable in speed.

The construction of the continuously variable transmission 1 will be described in detail starting with the first shaft 3.

The first shaft 3 is connected to an engine crank shaft 2 by a damper device 12 that absorbs engine torque fluctuation, and it forms the input shaft. The input shaft 3 is constituted by a stationary sheave 7a of the primary pulley 7 and a shaft 3a spline-fitted to a boss $7a_1$ of the stationary sheave 7a. The shaft 3a constituting the input shaft 3 is fixed to an input-side member 13 of a low clutch (running clutch) C_L, and supports an output-side member 15 of the low clutch C_L freely rotatably. The output member 15 is firmly connected to a primary-side sprocket 18 that constitutes a constant speed transmission apparatus 16. The stationary sheave 7a of the primary pulley 7 constituting the input shaft 3 is connected to an oil pump 17. The stationary sheave 7a supports a movable sheave 7b in such a manner that the movable sheave 7b is movable in the directions of the axis. The movable sheave 7b is axially moved by a first hydraulic actuator 7c.

The second shaft 4 is constituted by a stationary sheave 9a of the secondary pulley 9. The stationary sheave 9a supports a movable sheave 9b in such a manner that the movable sheave 9b is movable in the directions of the axis. The movable sheave 9b is axially moved by a second hydraulic actuator 9c provided as an axial force applying means. Disposed on the second shaft 4 are a high clutch C_H and the planetary gear 19. Further, a secondary-side sprocket 20 and an output gear 21 are freely rotatably supported by the second shaft 4.

The planetary gear 19 is constituted by a single pinion planetary gear made up of the sun gear 19s, the ring gear 19r and the carrier 19c freely rotatably supporting a pinion 19p meshed with the sun and ring gears. The sun gear 19s

is connected to the stationary sheave 9a of the secondary pulley 9 constituting the second shaft 4, and it forms the second rotary element. The ring gear 19r is connected to the output gear 21, and it forms the third rotary element. The carrier 19c is connected to the secondary-side sprocket 20, and it forms the first rotary element. The primary-side and secondary-side sprockets 18, 20 are wrapped around by a sprocket connector 22 such as a silent chain, a roller chain or a timing belt. The high clutch C_H is disposed between the sun gear 19s and the ring gear 19r.

The output gear 21 meshes with a large gear 23a of the countershaft 5 forming the third shaft. A small gear 23b of the shaft 5 meshes with a ring gear 24 of a differential 25. The differential 25 outputs differential rotations to the right and left axles 6a, 6b constituting the fourth shaft 6.

The arrangement and construction of the continuously variable transmission 1 will be described further in detail.

The boss $7a_1$ of the stationary sheave 7a of the primary pulley 7 supports the movable sheave 7b by means of a ball spline 30 in such a manner that the movable sheave 7b is freely movable in the direction of the axis. Disposed behind the movable sheave 7b is the first hydraulic actuator 7c described later. At the rear end side of the actuator 7c (the side remote from the engine crank shaft 2), the boss $7a_1$ is freely rotatably supported on a case 33 by a (ball) bearing 32. The stationary sheave 7a has a jaw $7a_2$ formed on its back face, and is freely rotatably supported by a (roller) bearing 35 disposed on the inner peripheral face of the jaw $7a_2$. Thus, the primary pulley 7 is supported at both ends by the aforementioned bearings 32, 35.

The oil pump 17 is disposed on the first shaft 3, between the damper device 12 and the stationary sheave 7a of the primary pulley 7. The primary-side sprocket 18 of the constant speed transmission apparatus 16 is disposed on the first shaft 3, at the rear end side of the bearing 32 of the primary pulley 7. The low clutch C_L is disposed on the rear end side of the primary-side sprocket 18. Thus, the oil pump 17, the stationary sheave 7a and the movable sheave 7b of the primary pulley 7, the first hydraulic actuator 7c, the bearing 32, the primary-side sprocket 18 and the low clutch C_L are arranged on the first shaft 3 in that order progressing from the engine side (base end side) to the rear end side of the shaft.

The boss $9a_1$ of the stationary sheave 9a of the secondary pulley 9 supports the movable sheave 9b by a ball spline 36 in such a manner that the movable sheave 9b is movable in the directions of the axis. Disposed behind the movable sheave 9a is the second hydraulic actuator 9c described later. The secondary pulley 9 is provided with a (ball) bearing 37 disposed on the outer peripheral face of the boss on the back side of the stationary sheave 9a, and a (roller) bearing 39 disposed on the outer peripheral face of the boss on the back side of the hydraulic actuator 9c. These bearings 37, 39 support the case 33 at both ends. The axial arrangements of the stationary and movable sheaves of the primary pulley 7 and the secondary pulley 9 are opposite to each other with respect to the belt 10. More specifically, the movable sheave 9b of the secondary sheave 9 and the hydraulic actuator 9c behind the movable sheave 9b are disposed on the engine side (base end side), and the stationary sheave 9a of the secondary sheave 9 is disposed on the forward end side.

The output shaft 21 is freely rotatably disposed on the second shaft 4, on the end side of the bearing 37 of the stationary sheave 9a, by a (ball) bearing 40. Disposed on the end side of the output shaft 21 is the high clutch C_H made up of an input-side member 41 fixed to the stationary sheave 9a of the secondary pulley 8 constituting the second shaft 4, and an output-side member 42 fixed to the output gear 21. The planetary gear 19 is disposed on the end side of the high clutch C_H. A hollow intermediate shaft 44 is freely rotatably supported on the end side of the planetary gear 19. The intermediate shaft 44 connects the secondary-side sprocket 20 of the constant speed transmission apparatus 16 to the carrier 19c of the planetary gear 19. Thus, the second hydraulic actuator 9c, the movable sheave 9b, the stationary sheave 9a, the output shaft 21, the high clutch C_H, the planetary gear 19, and the secondary sprocket 20 of the constant speed transmission apparatus 16 are arranged on the second shaft 4, in that order going from the engine side (base end side) to the rear end side of the shaft.

The hydraulic actuators 7c, 9c of the primary and secondary pulleys 7, 9 comprise cylinder members 47, 49 and partition members 45, 46 fixed to the stationary sheave bosses $7a_1$, $9a_1$, and second piston members 52, 53 and drum members 50, 51 fixed to the back sides of the movable sheaves 7b, 9b, respectively. The partition members 45, 46 are oil-tightly fitted to the second piston members 52, 53, and the second piston members 52, 53 are oil-tightly fitted to the cylinder members 47, 49 and the partition members 45, 46, thus forming double piston structures having a first hydraulic chamber 55 and a third hydraulic chamber 57, and a second hydraulic chamber 56 and a fourth hydraulic chamber 59, respectively.

In the first hydraulic chamber 55 and the second hydraulic chamber 56 of the hydraulic actuators 7c, 9c, the piston faces are formed by the back side faces of the movable sheaves 7b, 9b. The primary and secondary pulleys 7, 9 have an equal effective pressurized areas of their piston faces. The stationary sheave bosses $7a_1$, $9a_1$ of the primary and secondary pulleys have fluid passages connecting to the first hydraulic chamber 55, the second hydraulic chamber 56, the third hydraulic chamber 57, and the fourth hydraulic chamber 59. The primary pulley 7 and the secondary pulley 9 are provided with a pre-loading spring (first urging means) 65 and a pre-loading spring (second urging means) 66 that urge the movable sheaves 7b, 9b toward the stationary sheaves 7a, 9a, respectively.

The invention is characterized in that the loads (urging forces) of the pre-loading springs 65, 66 are suitably set, which feature will be described later.

The arrangement and construction of the components of the continuously variable transmission 1 will be briefly reviewed with reference to Figs. 1 and 2. As shown in Fig. 1, the damper device 12 and the oil pump 17 on the first shaft 3 overlap the second hydraulic actuator 9c of the secondary pulley 9 on the second shaft, in the axial directions. The planetary gear 19, the high clutch C_H and the output gear 21 on the second shaft 4, disposed between the stationary sheave 9a and the secondary-side sprocket 20 of the constant speed transmission apparatus 16, overlap the first hydraulic actuator 7c of the primary pulley 7 disposed on the first shaft 3, in the axial directions.

In a side view as shown in Fig. 2, the lines connecting the first shaft 3, the second shaft 4, the third shaft 5 and the fourth shaft 6 form a distorted quadrilateral. When installed in a vehicle, they are arranged in the order of the first, second, fourth and third shafts, starting from the front, and in the order of the second shaft 4, the third shaft 5, the first shaft 3 and the fourth shaft 6, starting from the top.

The foregoing arrangement and construction of the continuously variable transmission 1 advantageously achieves a size reduction of the entire system, particularly, a size reduction in the axial direction.

The operation of the above-described continuously variable transmission 1 will be described with reference to Figs. 1, 3 and 4. The clutch engagement table of Fig. 4 indicates clutch engagement by a symbol ○ and disengagement by blanks.

The revolution of the engine crank shaft 2 is transmitted to the input shaft 3 by the damper device 12. In a Low mode of the D (drive) range where the low clutch C_L is engaged and the high clutch C_H is disengaged, the rotation of the input shaft 3 is transmitted to the primary pulley 7, and also transmitted to the carrier 19c of the planetary gear 19 by the constant speed transmission apparatus 16 made up of the primary-side sprocket 18, the sprocket connector 22 and the secondary-side sprocket 20. The rotation of the primary pulley 7 is continuously changed in speed by suitably adjusting the pulley ratio (indicated by "CVT RATIO" in drawings) of the primary and secondary pulleys by using the hydraulic actuators 7c, 9c described later, and transmitted to the secondary pulley 9. The speed-changed rotation of the pulley 9 is transmitted to the sun gear 19s of the planetary gear 19.

In the planetary gear 19, the continuously speed-changed rotation from the belt-type continuously variable transmission apparatus (CVT) 11 is transmitted to the sun gear 19s while the carrier 19c is receiving the constant speed rotation from the constant speed transmission apparatus 16 and thereby serving as a reaction element, as indicated in the speed graph of Fig. 3. The resultant rotation of the rotations of the carrier 19c and the sun gear 19s is transmitted to the output gear 21 by the ring gear 19r. In this process, because the output shaft 4 is connected to the ring gear 19r, which is a rotary element other than the reaction force support element, the planetary gear 19 undergoes torque circulation, and the sun gear 19s and the carrier 19c rotate in the same direction, so that the output shaft can rotate in the forward (Low) and reverse directions, intervened by zero-rotation. In accordance with the aforementioned torque circulation, torque is transmitted from the secondary pulley 9 to the primary pulley 7 of the CVT 11 while the output shaft 4 is rotating in the forward direction (forward drive), and torque is transmitted from the primary pulley 7 to the secondary pulley 9 while the output shaft 4 is rotating in the reverse direction (reverse drive).

In a High mode where the low clutch C_L is disengaged and the high clutch C_H is engaged, the torque transmission to the planetary gear 19 through the constant speed transmission apparatus 16 is disconnected, and the planetary gear 19 becomes an integrally rotating state due to the engagement of the high clutch C_H. Thus the rotation of the input shaft 3 is transmitted to the output gear 21 by the CVT 11 and the high clutch C_H. That is, the CVT 11 transmits drive force from the primary pulley 7 to the secondary pulley 9. The rotation of the output gear 21 is transmitted to the differential 25 by gears 23a, 23b of the countershaft 5, and then to the right and left front wheels by the right and left axles 6a, 6b.

As indicated in the speed graph of Fig. 3, the output torque graph of Fig. 5, and the output rotational speed graph of Fig. 6, operation is performed in the Low mode as follows. If the pulley ratio (CVT RATIO) of the CVT 11 is at the limit (O/D limit) in the speed increase direction (overdrive) (in a position indicated by the solid line a in Fig. 3), the ring gear 19r reversely rotates in response to the constant speed rotation of the carrier 19c and transmits the reverse rotation (REV) to the output gear 21 because the sun gear 19s rotates at the maximum speed in this situation. If the CVT 11 is shifted in the speed decrease direction (underdrive), the reverse rotation slows and, when the pulley ratio reaches a ratio determined by the gear ratios of the planetary gear 19 and the constant speed transmission apparatus 16, the CVT 11 assumes the neutral position (NEU) where the rotation of the output gear 21 becomes null. If the CVT 11 is further shifted in the speed decrease direction, the rotation of the ring gear 19r is switched to the forward direction and the forward rotation, that is, forward drive rotation, is transmitted to the output gear 21. As indicated in the graph of Fig. 5, the torque of the output gear 21 diverges to infinite in the vicinity of the neutral position NEU.

If the CVT 11 is shifted to the speed decrease (U/D) direction limit, the high clutch C_H engages to switch to the High mode. In the High mode, the output rotation of the CVT 11 is directly transmitted to the output gear 21 as indicated by the broken line b parallel to the abscissa axis of the speed graph of Fig. 3. Then as the CVT 11 is shifted in the speed increase (O/D) direction, the rotation of the output gear 21 switches to the speed increase (O/D) direction while the transmitted torque correspondingly decreases. The symbol λ in Fig. 3 represents a ratio (Zs/Zr) of the number Zs of teeth of the sun gear and the number Zr of teeth of the ring gear.

In the P (parking) range and the N (neutral) range shown in the clutch engagement table of Fig. 4, both the low clutch C_L and the high clutch C_H are disengaged, and the transmission of the drive force from the engine (E/G) is dis-

connected. In the P range, the differential 25 is locked, and the axles 6a, 6b are also locked.

In the Low mode of the D range, the fluid pressure based on the oil pump 17 is supplied to the hydraulic servo for the low clutch C_L, so that the low clutch C_L engages. At the same time, the axial force produced by the second hydraulic actuator 9c of the secondary pulley 9 of the CVT 11 receiving fluid pressure in both the second and fourth hydraulic chambers 56, 59 becomes greater than the axial force by the first hydraulic actuator 7c receiving fluid pressure only in the first hydraulic chamber 55, so that the torque transmission occurs from the secondary pulley 9 to the primary pulley 7. By suitably adjusting the operation of the ratio control valve in that axial force condition of the pulleys 9, 7 corresponding to the aforementioned torque transmission, the fluid pressure in the fourth hydraulic chamber 59 of the second hydraulic actuator 9c is adjusted, so that axial force by the hydraulic actuator 9c is suitably adjusted, thus suitably changing the pulley ratio (torque ratio). The engine torque transmitted to the carrier 10c of the planetary gear 19 by the input shaft 3, the low clutch C_L and the constant speed transmission apparatus 16 in this condition is restricted by the sun gear 19s in the CVT 11 in accordance with the predetermined pulley ratio, and then extracted from the output gear 21 via the ring gear 19r.

In the High mode of the D range, a predetermined fluid pressure based on the oil pump 17 is supplied to the first and third hydraulic chambers 55, 57 of the first hydraulic actuator 7c of the primary pulley 7, and also supplied to the second hydraulic chamber 56 of the second hydraulic actuator 9c of the secondary pulley 9, and also supplied to the high clutch hydraulic servo. That is, although, in the High mode, the CVT 11 is in the D range position as in the Low mode, the high clutch C_H engages, and the axial force produced by the first hydraulic actuator 7c of the primary pulley 7 of the CVT 11 receiving fluid pressure in both the first and third hydraulic chambers 55, 57 becomes greater than the axial force by the second hydraulic actuator 9c receiving fluid pressure only in the second hydraulic chamber 56. By suitably adjusting the ratio control valve in the axial force condition corresponding to the torque transmission from the primary pulley 7 to the secondary pulley 9, the fluid pressure in the third hydraulic chamber 57 of the first hydraulic actuator 7c of the primary pulley 7 is adjusted, so that axial force by the hydraulic actuator 7c is suitably adjusted, thus achieving a suitable pulley ratio (torque ratio). The engine torque transmitted to the input shaft 3 in this condition is suitably changed by the CVT 11 transmitting from the primary pulley 7 to the secondary pulley 9, and then extracted from the output gear 21 via the high clutch C_H.

In the R (reverse) range, a predetermined hydraulic pressure is supplied to the first and third hydraulic chambers 55,57, and it is also supplied to the third hydraulic chamber 56 of the second actuator 9c of the secondary pulley 9 and to the low clutch hydraulic servo. Thereby the low clutch C_L engages, and the axial force produced by the first hydraulic actuator 7c of the primary pulley 7 of the CVT 11 receiving fluid pressure in both the first and third hydraulic chambers 55, 57 becomes greater than the axial force by the second hydraulic actuator 9c receiving fluid pressure only in the second hydraulic chamber 56, thus reaching an axial force condition corresponding to the torque transmission from the primary pulley 7 to the secondary pulley 9. By adjusting the ratio control valve, the fluid pressure in the third hydraulic chamber 57 of the first hydraulic actuator 7c of the primary pulley 7 is adjusted, so that axial force by the hydraulic actuator 7c is suitably adjusted, thus achieving a suitable pulley ratio (torque ratio). In this state, the pulley ratio of the CVT 11 is in a predetermined speed increase (O/D) state, and the engine torque from the input shaft 3 is transmitted to the carrier 19c of the planetary gear 19 by the low clutch C_L and the constant speed transmission apparatus 16, and also to the sun gear 19s by the CVT11 transmitting torque from the primary pulley 7 to the secondary pulley 9. The two torques are combined by the planetary gear 19, and the resultant torque is outputted as reverse rotation from the ring gear 19r to the output gear 21.

Finally described will be the features of the invention, the setting of the urging forces of the first and second urging means of the axial force applying means in the continuously variable transmission 1 having the foregoing construction and functions, more specifically, the setting of the spring loads of the first spring 65 of the planetary side and the second spring 66 of the secondary side.

The continuously variable transmission 1 performs neutral control when the vehicle is taking off.

The neutral control during takeoff is necessary because of the following circumstances. The continuously variable transmission 1 described above transmits the engine torque from the engine output shaft 2 to the input shaft 3 merely by the damper device in a direct manner, thus achieving a simple construction by omitting the conventionally required takeoff devices, such as a torque converter, a fluid coupling, an electromagnetic powder clutch, an input clutch, etc. To enable this construction, it is necessary for the continuously variable transmission 1 to perform so-called neutral control that converges the pulley ratio to a ratio that achieves the neutral state of the continuously variable transmission 1 and thereby prevents torque transmission to the output shaft 4 when a vehicle is stopped in the Low mode of the D range, where the low clutch C_L is engaged.

For takeoff, the engine is first started in the P or N range (vehicle stop range). Thus the oil pump 17 supplies equal fluid pressure to the first hydraulic chamber 55 of the first hydraulic actuator 7c of the primary pulley 7 and the second hydraulic chamber 56 of the second hydraulic actuator 9c of the secondary pulley 9, so that the movable sheaves 7b, 9b move toward the stationary sheaves 7a, 9a and clamp the belt 10. The movable sheaves 7b, 9b are pre-urged toward the stationary sheaves 7a, 9a by the first and second springs 65, 66. The forces of the fluid pressure and the springs 65, 66 balance the pulleys 7, 9 at the predetermined pulley ratio r_1. Then, by shifting from the vehicle stop range to the

D range (Low mode) for takeoff, the low clutch C_L engages and the neutral control starts so that the pulleys 7, 9 become steady at a pulley ratio r_N corresponding to the neutral state (the pulley ratio $r_N < 1$ according to this embodiment). The continuously variable transmission 1 thus become ready for takeoff. By depressing the accelerator pedal, fluid pressure is supplied to the fourth hydraulic chamber of the secondary pulley so that the pulley ratio shifts from the pulley ratio r_N for the neutral state toward the U/D side as indicated in Fig. 6. As a result, the continuously variable transmission 1 is shifted to the O/D side for acceleration.

The pulley ratio r_N corresponding to the neutral state is specifically determined by the construction of the planetary gear 19, the speed ratio of the constant speed transmission apparatus 16, etc. The detail of the neutral control is described in Japanese patent application Nos. HEI-7-66234 and HEI-7-128701.

By the neutral control during the shift from the vehicle stop range to the D range (Low mode) for vehicle takeoff, the pulley ratio is changed from the ratio r_1 to the pulley ratio r_N for the neutral state before actually taking off the vehicle. Therefore, if the change from the pulley ratio r_1 to the pulley ratio r_N requires a relatively long time, a response delay will occur in vehicle takeoff.

The invention prevents a takeoff response delay by reducing the time required for the pulley ratio change. As a means for the time reduction, the invention suitably sets the spring loads of the first and second pre-loading springs 65, 66 in a manner described later.

Figs.7, 8 illustrate different embodiments of the load setting of the springs. According to the embodiment illustrated in Fig. 7, the loads of the springs 66, 67 are set so that the pulley ratio (1) occurring before the first and second hydraulic chambers 55, 56 are supplied with minimum fluid pressures (minimum required fluid pressures) is slightly greater than the pulley ratio r_N corresponding to the neutral state. In contrast, according to the embodiment illustrated in Fig. 8, the loads of the springs 66, 67 are set so that the pulley ratio (2) occurring after the first and second hydraulic chambers 55, 56 are supplied with minimum fluid pressures is slightly greater than the pulley ratio r_N for the neutral state.

These embodiments will be described in detail.

The O-ring sliding resistance is first determined (as indicated by the straight line a in Fig. 7). O-rings 7d, 9d are disposed between the outer peripheral face of the boss $7a_1$ of the stationary sheave 7a and the inner peripheral face of the movable sheave 7b of the primary pulley 7, and between the outer peripheral face of the boss $9a_1$ of the stationary sheave 9a and the inner peripheral face of the movable sheave 9b of the secondary pulley 9, as shown in Fig. 1. The O-rings 7d, 9d provide resistance when the movable sheaves 7b, 9b axially move to change the pulley ratio.

Next determined will be the pulley minimum required load to prevent the belt 10 from slipping immediately after the engine is started, as indicated by the curve b in Fig. 7. Below this load, the belt may fall off, or insufficient drive force transmission may result.

The spring load of the spring 66 of the secondary pulley 9 is set so that the load is always above the lines a and b (as indicated by the curve c). If the spring load of the spring 66 is excessively large, it will make a great resistance that impedes the convergence of the pulley ratio to the pulley ratio r_N for the neutral state. Therefore, the load of the spring 66 should preferably be set to a minimum possible level. The load of the spring 65 of the primary pulley 7 is set in the same manner.

Then the spring load of the primary pulley 7 that causes the primary pulley 7 to balance with the secondary pulley 9 before the first and second hydraulic chambers 55, 56 are supplied with minimum fluid pressure is determined (the curved) under the conditions where the secondary-side spring 66 is mounted but the primary-side spring 65 is not mounted. In a similar manner, the primary-side spring load to cause the primary pulley 7 to balance with the secondary pulley 9 after the first and second hydraulic chambers 55, 56 have been supplied with minimum fluid pressure is determined (the curve e).

Finally, the spring load of the primary-side spring 65 (the curve f) is determined with reference to the curves d, e in the following manner.

As a precondition for the determination, the spring load of the primary-side spring 65 must be set greater than the spring load of the secondary-side spring 66 in order to establish a pulley ratio approximate to the pulley ratio r_N (< 1) for the neutral state (that is, to shift the ratio toward the O/D side). In addition, the embodiment illustrated in Fig. 7 sets the spring loads in accordance with the following relation between the pulley ratio (1) before supply of the minimum fluid pressure and the pulley ratio (2) after supply of the minimum fluid pressure:

$$\text{pulley ratio } r_N < \text{pulley ratio (1)} < \text{pulley ratio (2)} \qquad \text{formula 1}$$

The pulley ratio (1) is set slightly greater for the following reasons. Drives are expected to shift to the D range more often than to the R range, immediately after start of the engine, that is, before a fluid pressure rise. In addition, if pulley ratio (1) = pulley ratio r_N is set, errors in the processing and assembly of the springs 65, 66 may result in the actual pulley ratio (1) < the pulley ratio r_N after the assembly. The setting of pulley ratio r_N < pulley ratio (1) will prevent such a consequence, thus preventing at least reverse drive of the vehicle.

It is expected that the spring load of the primary-side spring 65 that satisfies the precondition and formula 1 will follow a curve symmetric to the spring load (curve c) of the secondary-side spring 66, with respect to a lateral axis.

Although the value of the load is not determined yet, the curve is expected to become like the curve f. The curve f is shifted along the ordinate axis and the abscissa axis to determined its position. The intersection of the curve f and the curved represents the pulley ratio (1), and the intersection of the curve f and the curve e represents the pulley ratio (2). To summarize, the conditions are: the curve c < the curve f; the curves c, f are minimum possible; the pulley ratio (1) and the pulley ratio (2) are within the range between the pulley ratio r_N and the pulley ratio 1; and the pulley ratio (1) is closer to the pulley ratio r_N than the pulley ratio (2) is close to the pulley ratio r_N.

The spring load of the primary-side spring 65 is determined so that these conditions are satisfied.

According to the embodiment illustrated in Fig. 7, since the pulley ratio (1) before supply of the minimum fluid pressure becomes approximate to the pulley ratio r_N for the neutral state solely on the basis of the load setting of the springs 65, 66, it is possible to prevent the vehicle from taking off backward against the intention of the driver even if the driver shifts from the vehicle stop range to the D range (Low mode) for takeoff immediately after starting the engine, that is, before a fluid pressure rise.

When the fluid pressure rises to a sufficiently high level, the pulleys balance at the pulley ratio (2). Since the spring loads of the springs 65, 67 are pre-set so that the pulley ratio converges to the neutral-state pulley ratio r_N within the period between the start and the completion of engagement (connection) of the low clutch (running clutch) C_L, the embodiment prevents a takeoff response delay caused by the change from the pulley ratio (2) to the pulley ratio r_N.

Although this embodiment sets the spring load of the secondary-side spring 66 prior to the spring load of the primary-side spring 65, this setting sequence may be reversed. The reversed sequence is able to determine the spring load of the secondary-side spring 66 in substantially the same procedures as described above. However, since the reversed sequence will determine the curve c below the curve f and above the curves a and b, the requirements for determining the curve c will increase over those for determining the curve f according the embodiment of Fig. 7, thus resulting in more time and labor for the setting.

The embodiment illustrated in Fig. 8 will be described.

This embodiment sets the spring loads so that the pulley ratio (2) after supply of the minimum fluid pressure will become slightly greater than the pulley ratio r_N for the neutral state, while the embodiment of Fig. 7 sets so that the pulley ratio (1) before supply of the minimum fluid pressure will become slightly greater than the neutral-state pulley ratio r_N. That is, the embodiment of Fig. 8 determines the spring load of the primary-side spring 65 so that the following condition is satisfied:

$$\text{pulley ratio (1)} < \text{pulley ratio } r_N < \text{pulley ratio (2)}$$

The embodiment of Fig. 8 determines the curves a-e in the same manner as in the embodiment of Fig. 7, but determines the curve f in a different manner.

The embodiment of Fig. 8 determines such a position of the curve f that pulley ratio (2) defined by the intersection of the curve f and the curve e becomes as great as possible. Since this determination manner makes it impossible to obtain an intersection of the curve f and the curve d, this embodiment sets the pulley ratio (1) to a value approximate to the pulley ratio at the far-end limit in the O/D side.

Although the pulley ratio (1) < the pulley ratio r_N, that is, a pulley ratio corresponding to the reverse drive, is maintained before a fluid pressure rise, the embodiment will achieve the pulley ratio (2), that is, the target pulley ratio slightly greater than the pulley ratio r_N, after the fluid pressure rise. Practically, it is rare for drivers to shift from the vehicle stop range to the D range (Low mode) for takeoff immediately after starting the engine, that is, before the fluid pressure rise. In addition, even in such a rare case, backward takeoff can be prevented by controlling the engage timing of the low clutch C_L. Since in normal takeoff operation, drivers usually shift the range after a sufficient fluid pressure rise following start of the engine, it may be considered that this embodiment, using the pulley ratio (2) after the minimum fluid pressure supply being approximate to the pulley ratio r_N, is more suitably adapted to practical takeoff operations than the embodiment of Fig. 7.

Since the spring loads of the primary-side and secondary-side springs 65, 66 are set such that the pulley ratios (1), (2) before and after the minimum fluid pressure supply become approximate to the pulley ratio r_N according to the embodiment of Fig. 7, or such that the pulley ratio (2) after the minimum fluid pressure supply becomes approximate to the pulley ratio r_N according to the embodiment of Fig. 8, it is possible to reduce the time required for the pulley ratio shift to the neutral-state pulley ratio r_N during the takeoff operation. The embodiments thus effectively prevent the vehicle takeoff response delay.

As described above, the gist of the invention is to reduce the time required for the pulley ratio shift to the pulley ratio r_N corresponding to the neutral state during the vehicle takeoff operation and, thereby, eliminate the takeoff response delay by setting the urging forces (spring loads) of the first urging means (the first spring 65) and the second urging means (the second spring 66). The setting method is not limited to the embodiments illustrated in Figs. 7, 8, but other methods may be employed. For example, a method may be employed which simultaneously disposes in the pulleys 7, 9 a pair of a primary-side spring 65 and a secondary-side spring 66 whose spring load has been set less than that of the primary-side spring 65, examines a plurality of such pairs of springs as to whether the requirements of the pulley

ratios (1), (2) are met, and determines the optimal pair to use.

The invention suitably sets spring loads of the springs 65, 66 such that the pulley ratio (1) based on the spring loads alone becomes approximate to the neutral-state pulley ratio r_N according to the embodiment of Fig. 7, or such that the pulley ratio (2) based on the spring loads and the fluid pressure combined becomes approximate to the neutral-state pulley ratio r_N. The value of the neutral-state pulley ratio r_N does not limit the invention.

While the invention has been described with reference to what are presently considered to be preferred embodiments thereof, it is to be understood that the invention is not limited to the disclosed embodiments. To the contrary, the invention is intended to cover various modifications and equivalent arrangements included within the spirit and scope of the appended claims.

According to the construction, the continuously variable transmission is able to quickly reach the pulley ratio r_N corresponding to the neutral state changing from the pulley ratio r_1 in a case where it is necessary to establish the pulley ratio r_N corresponding to the neutral state at the time of takeoff of a vehicle. Thus the invention reduces the response delay of the vehicle during taking-off caused by a change in the pulley ratio, enabling a smooth takeoff.

According to the construction, the continuously variable transmission is a simple construction in which the respective movable sheaves of the first and second pulleys are moved by the appropriate urging forces of the first and second springs.

According to the construction, the continuously variable transmission is able to achieve two pulley ratios r_1 and r_N in accordance with the presence of pressure supply to the first and second hydraulic chambers.

According to the construction, the continuously variable transmission sets the pulley ratio occurring before the two hydraulic chambers are supplied with minimum required fluid pressure, that is, one of the two pulley ratios, to the pulley ratio r_1. Therefore, for example, in a case where a driver shifts the range immediately after starting the engine (that is, before the hydraulic chambers are supplied with the minimum required pressure), the continuously variable transmission is able to hold the vehicle stop state, thus preventing movements of the vehicle that are not intended by the driver.

According to the construction, the continuously variable transmission is based on an expectation that drivers more often will perform a shift from a vehicle stop range (the P range or the N range) to the D range than a shift therefrom to the R range, immediately after starting the engine, and an expected possibility that the pulley ratio may fail to become approximate to the pulley ratio r_N corresponding to the neutral range because of errors in assembly or processing of the springs. By setting a pulley ratio r_1 biased to the forward drive state side, the continuously variable transmission is able to prevent rearward movements of the vehicle that are not intended by the driver.

According to the construction, the pulley ratio r_N is achieved before the completion of engagement of the running clutch, the continuously variable transmission enables a quick takeoff of the vehicle.

According to the construction, the continuously variable transmission sets the pulley ratio occurring after the two hydraulic chambers are supplied with the minimum required fluid pressures, that is, the other one of the two pulley ratios as described above, to the aforementioned pulley ratio r_1, so that the pulley ratio r_1 can be brought approximately to the pulley ratio r_N with a high precision. Thus the continuously variable transmission is able to further quicken the shift from the pulley ratio r_1 to the pulley ratio r_N, that is, it is able to very quickly achieve the neutral state when a driver shifts from the P range or the like to the D (drive) range.

Claims

1. A continuously variable transmission comprising:

    an input shaft (3) drivingly connected to an engine output shaft (2);
    an output shaft (4) drivingly connected to a wheel;
    a belt-type continuously variable transmission apparatus (11) having a first pulley (7) drivingly connected to said input shaft (3), a second pulley (9) provided on the side of the output shaft (4), a belt (10) wound around said two pulleys (7,9), and axial force applying means (7c,9c) for applying axial force to said pulleys (7,9) to change the pulley ratio; and
    a planetary gear (19) having a first rotary element (19c) connected to said input shaft (3), a second rotary element (19s) connected to said second pulley (9), and a third rotary element (19r) connected to said output shaft (4) wherein the pulley ratio being continuously changed to correspondingly achieve a forward drive state, a neutral state, and a reverse drive state of a vehicle through said output shaft (4),

    wherein said axial force applying means (7c,9c) having first urging means (65) for constantly urging said first pulley (7) to apply an axial force thereto, and second urging means (66) for constantly urging said second pulley (9) to apply an axial force thereto, and

    each of urging forces of said first and second urging means (65,66) being set such that a pulley ratio (r_1) becomes approximate to a pulley ratio r_N , where said pulley ratio r_1 is a pulley ratio achieved by said axial force applying means (7c,9c) in a vehicle stop range in which no torque is transmitted to said output shaft (4), and said

pulley ratio (r_N) is a pulley ratio that achieves said neutral state.

2. A continuously variable transmission according to claim 1,

wherein each of said first and second pulleys (7,9) comprises a stationary sheave (7a,9a) and a movable sheave (7b,9b) that clamp said belt (10) therebetween, and

wherein said first and second urging means (65,66) are a first spring (65) and a second spring (66), respectively, that force said movable sheaves (7b,9b) to clamp said belt (10).

3. A continuously variable transmission according to claim 2,

wherein said axial force applying means (7a,9a) having a first hydraulic chamber (55) that receives a fluid pressure supply to urge said movable sheave (7b) in cooperation with said first spring (65); and

a second hydraulic chamber (56) that receives a fluid pressure supply to urge said movable sheave (9b) in cooperation with said second spring (66).

4. A continuously variable transmission according to claim 3,

wherein each of urging forces of said first and second springs (65,66) are set such that said pulley ratio r_1 becomes approximate to said pulley ratio r_N before each of said first and second hydraulic chambers (55,56) is supplied with a minimum required fluid pressure.

5. A continuously variable transmission according to claim 4,

wherein said pulley ratio r_1 is a pulley ratio that achieves the forward drive state.

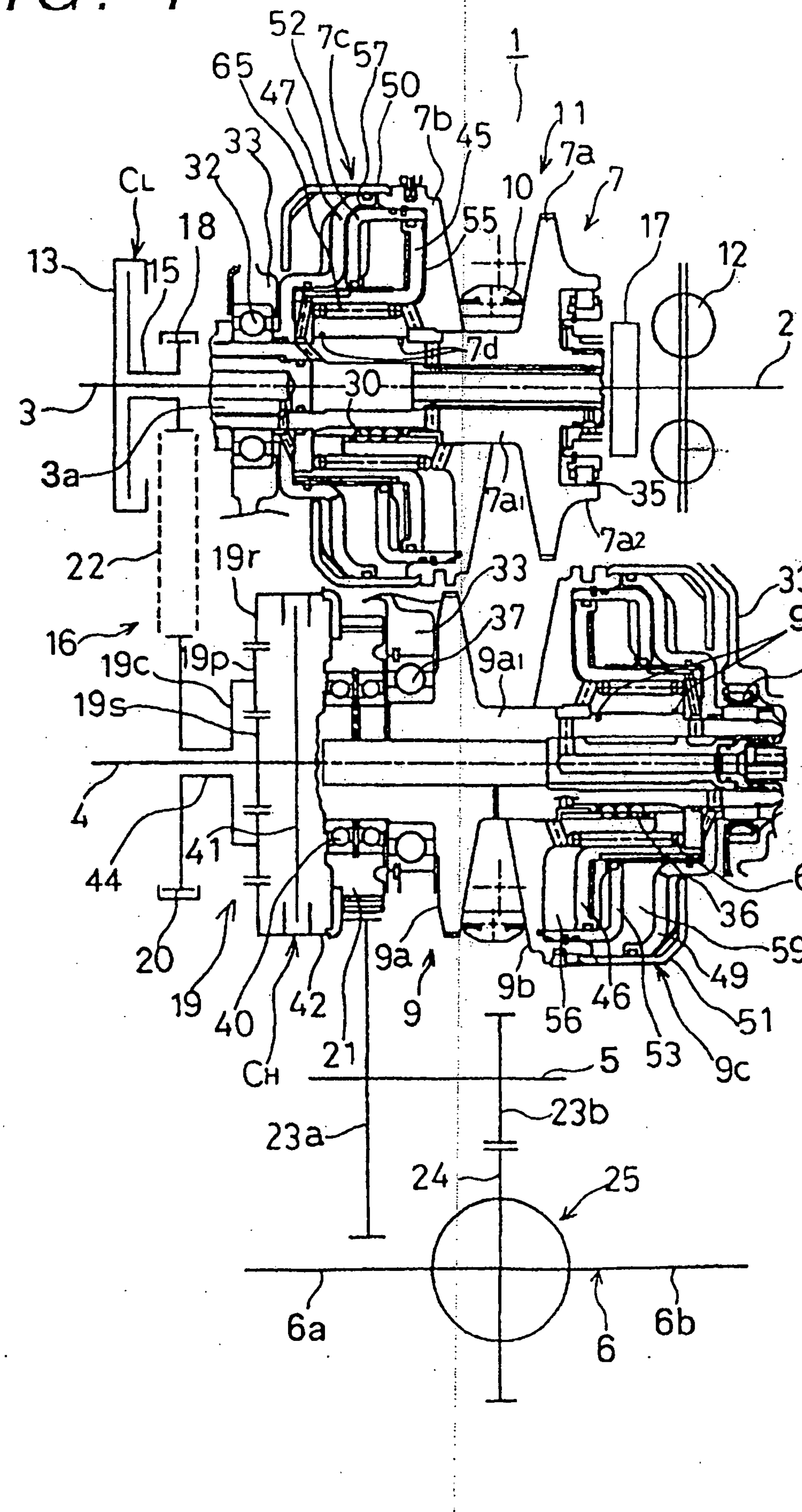
6. A continuously variable transmission according to claim 4, further comprising a running clutch (C_L) that is engaged in a vehicle driving range,

each of urging forces of said first and second springs (65,66) being set such that within a period between the start and the completion of engagement of said running clutch (C_L), the pulley ratio occurring after each of said first and second hydraulic chamber (55,56) is supplied with fluid pressure converges to said pulley ratio r_N.

7. A continuously variable transmission according to claim 3 or 4,

wherein each of urging forces of said first and second springs (65,66) are set such that said pulley ratio (r_1) becomes approximate to said pulley ratio (r_N) after each of said first and second hydraulic chambers (55,56) is supplied with a minimum required fluid pressure.

FIG. 1


1
11
7
7a
7b
7c
7d
10
12
13
15
17
18
2
3
3a
CL
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47
52
57
50
45
55
30
35
7a1
7a2
22
16
19r
19c
19p
19s
4
44
41
20
19
40
42
CH
21
9a
9
9b
9a1
9d
37
39
66
36
59
49
51
9c
53
46
56
5
23a
23b
24
25
6a
6
6b

FIG. 2

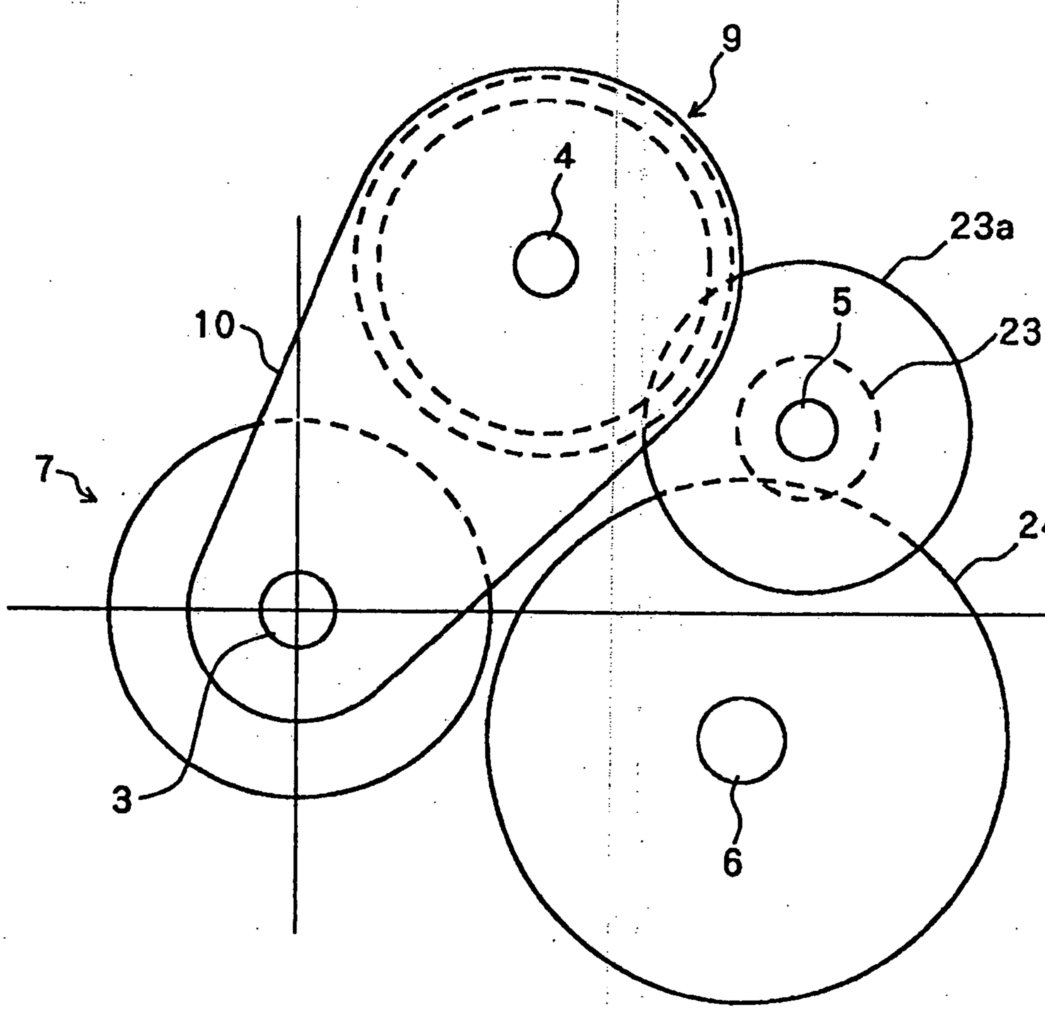


FIG. 3

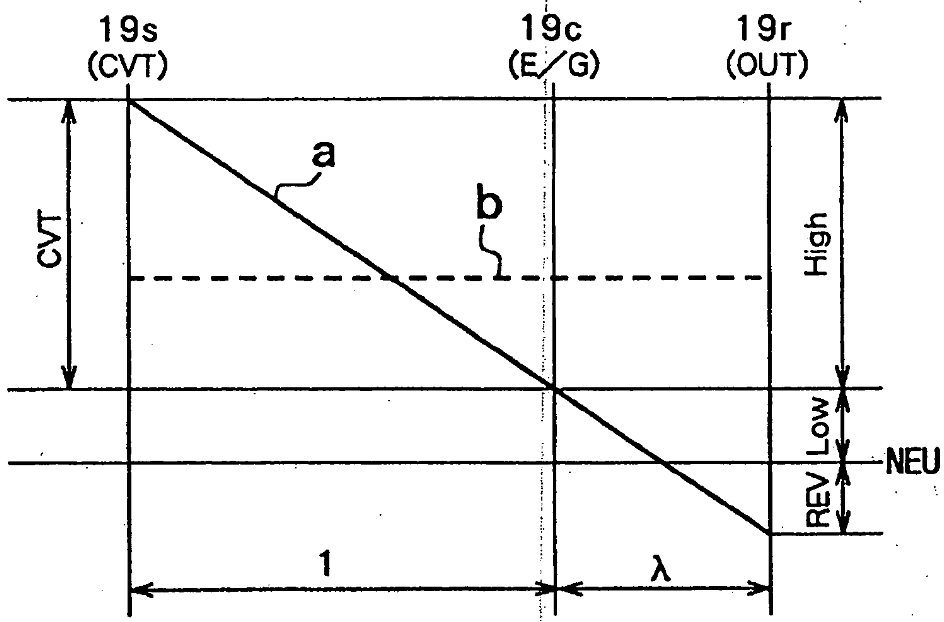
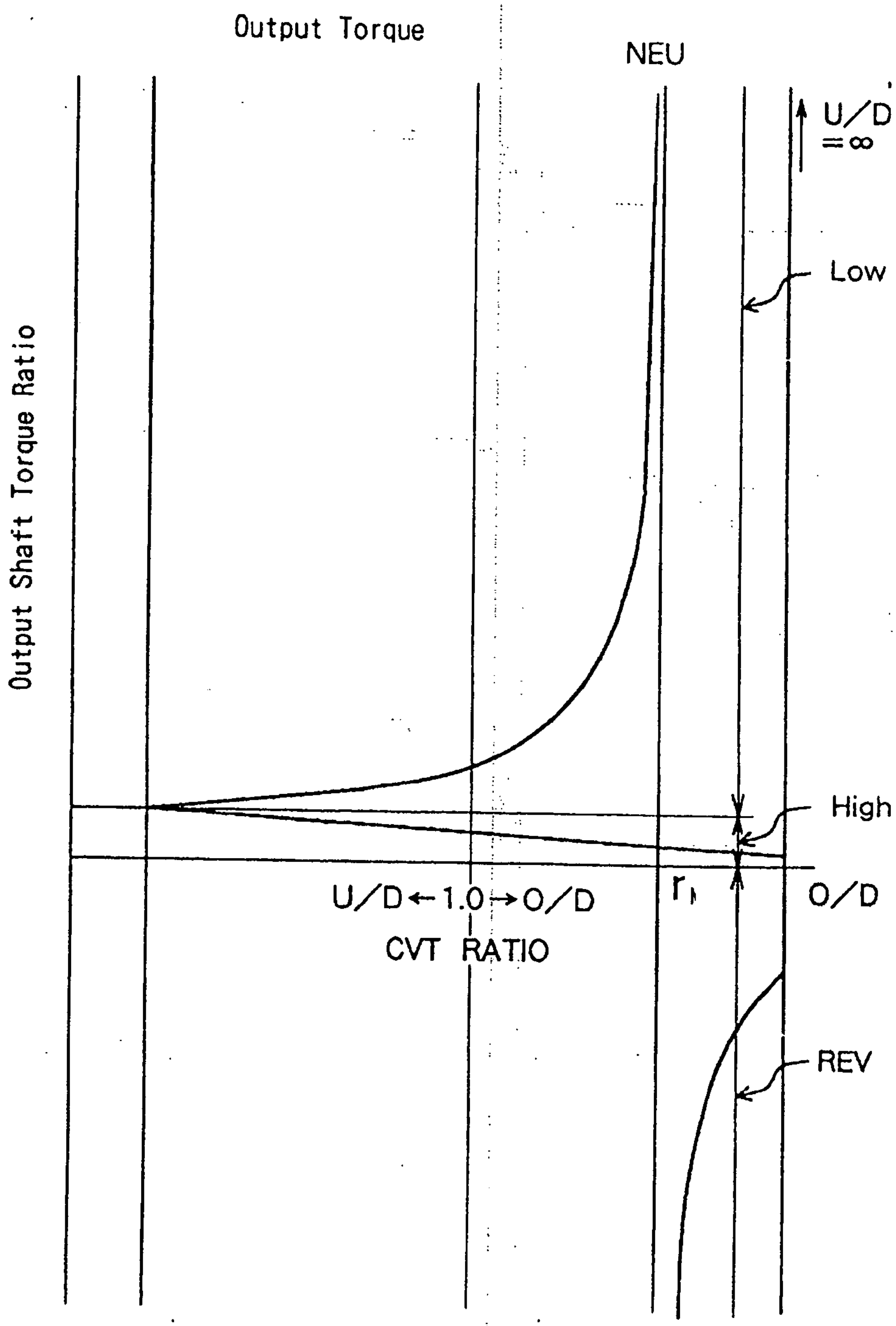


FIG. 4

Clutch Engagement Table

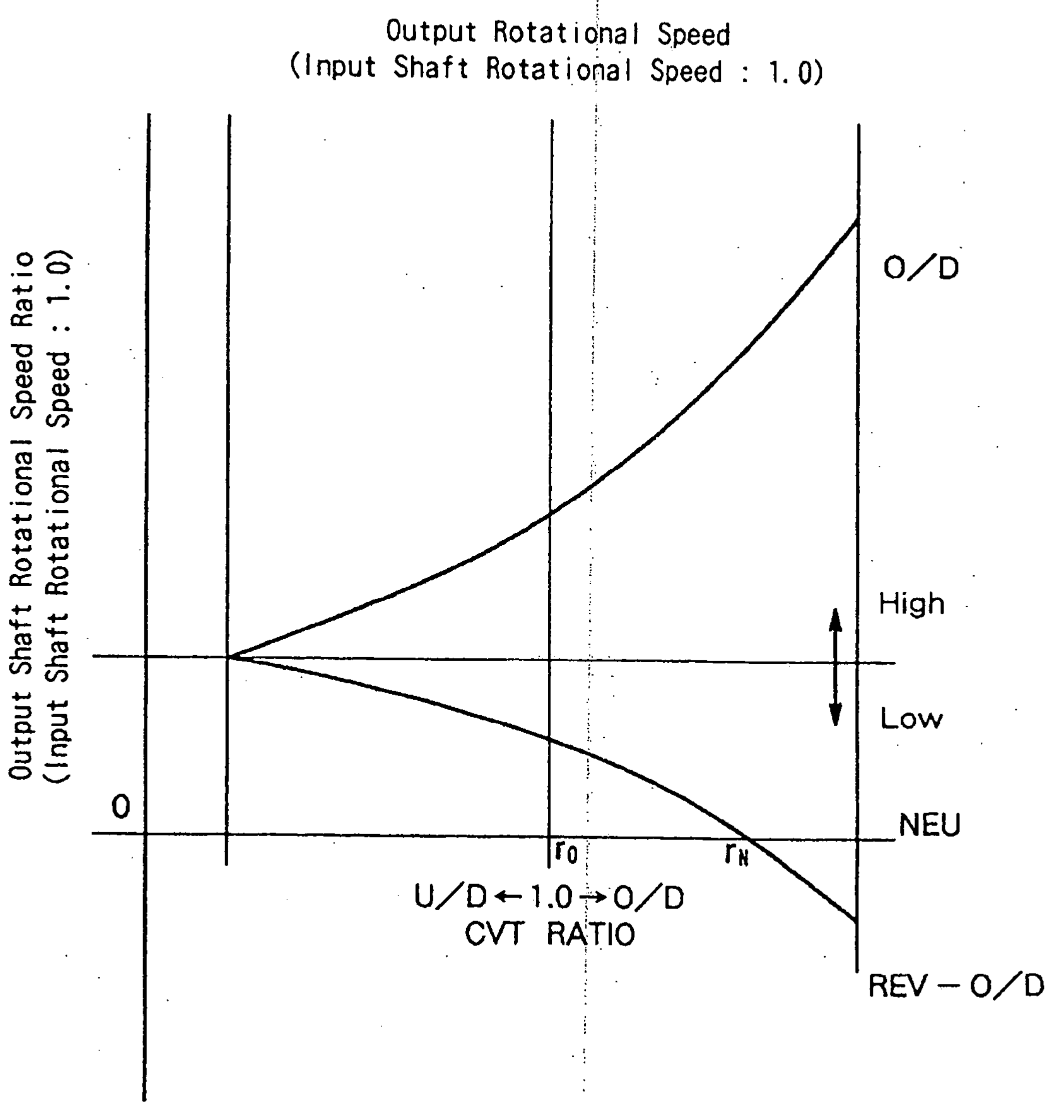
Range \ Clutch		C_L	C_H
P			
R		○	
N			
D	Low	○	
	High		○

FIG. 5



Output Torque
NEU
U/D =∞
Low
Output Shaft Torque Ratio
High
U/D←1.0→O/D
CVT RATIO
r
O/D
REV

FIG. 6


Output Rotational Speed
(Input Shaft Rotational Speed : 1.0)
Output Shaft Rotational Speed Ratio
(Input Shaft Rotational Speed : 1.0)
O/D
High
Low
NEU
REV – O/D
0
r_0
r_N
U/D←1.0→O/D
CVT RATIO

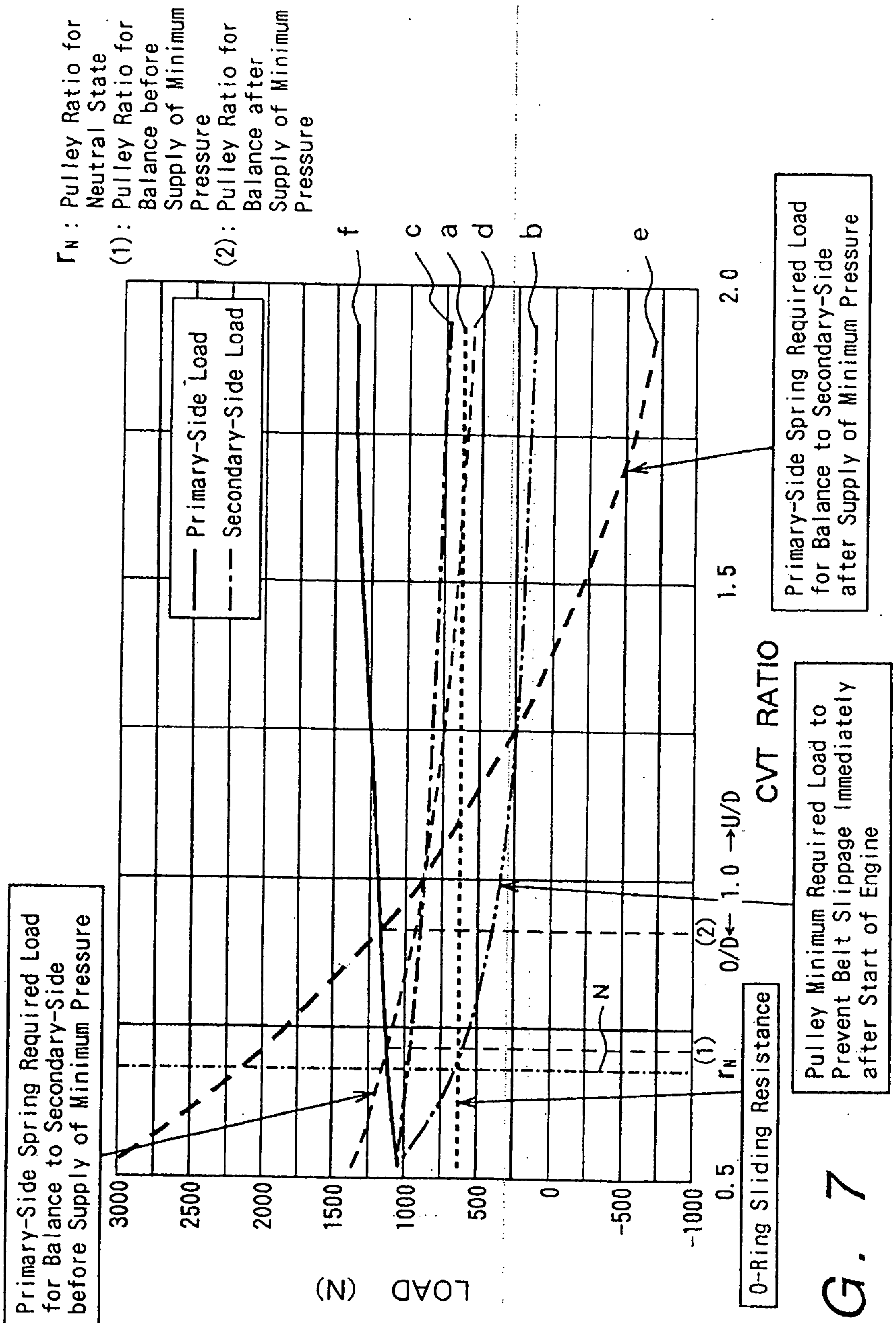
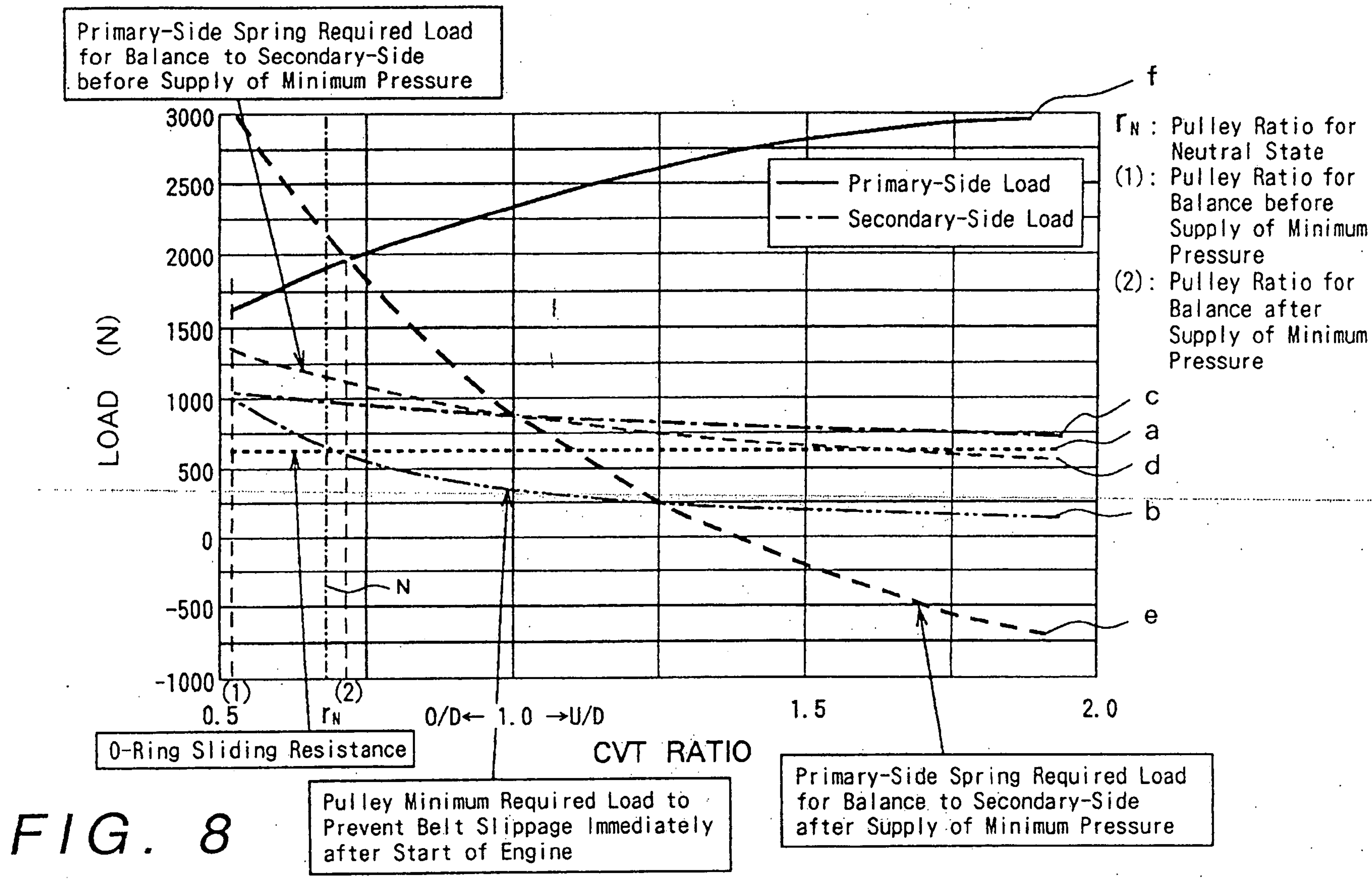

r_N : Pulley Ratio for Neutral State
(1) : Pulley Ratio for Balance before Supply of Minimum Pressure
(2) : Pulley Ratio for Balance after Supply of Minimum Pressure
Primary-Side Load
Secondary-Side Load
f
c
a
d
b
e
LOAD (N)
3000
2500
2000
1500
1000
500
0
-500
-1000
0.5
r_N
(1)
N
(2)
O/D←
1.0 →U/D
1.5
2.0
CVT RATIO
Primary-Side Spring Required Load for Balance to Secondary-Side before Supply of Minimum Pressure
O-Ring Sliding Resistance
Pulley Minimum Required Load to Prevent Belt Slippage Immediately after Start of Engine
Primary-Side Spring Required Load for Balance to Secondary-Side after Supply of Minimum Pressure

FIG. 7



Primary-Side Spring Required Load for Balance to Secondary-Side before Supply of Minimum Pressure
f
r_N : Pulley Ratio for Neutral State
(1): Pulley Ratio for Balance before Supply of Minimum Pressure
(2): Pulley Ratio for Balance after Supply of Minimum Pressure
Primary-Side Load
Secondary-Side Load
3000
2500
2000
1500
1000
500
0
-500
-1000
LOAD (N)
c
a
d
b
e
N
(1)
(2)
0.5
r_N
O/D← 1.0 →U/D
1.5
2.0
CVT RATIO
O-Ring Sliding Resistance
Pulley Minimum Required Load to Prevent Belt Slippage Immediately after Start of Engine
Primary-Side Spring Required Load for Balance to Secondary-Side after Supply of Minimum Pressure

FIG. 8

European Patent Office

EUROPEAN SEARCH REPORT

Application Number
EP 96 11 9290

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int.Cl.6)
Y	US 4 990 127 A (ROBERTS DECEASED RICHARD W ET AL) * column 7, line 4 - column 9, line 11; figure 1 *	1-4,6,7	F16H37/08
Y	EP 0 370 040 B (PERRY FORBES G D) * column 5, line 23 - column 6, line 31; claim 5 *	1-4,6,7	
X	GB 2 010 423 A (LELY NV C VAN DER) * page 5, line 41-121; figure 2 *	1-4,6,7	
X	US 3 131 581 A (GRAYBILL) * column 3, line 18-33 *	1,2	
X	DE 44 35 779 A (BANDO CHEMICAL IND) * column 13, line 12 - column 17, line 33 *	1	
A	PATENT ABSTRACTS OF JAPAN vol. 012, no. 109 (M-682), 8 April 1988 & JP 62 237166 A (DAIHATSU MOTOR CO LTD), 17 October 1987, * abstract *	5	

TECHNICAL FIELDS SEARCHED (Int.Cl.6)
F16H

The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
BERLIN	18 March 1997	Hunt, A

CATEGORY OF CITED DOCUMENTS

X : particularly relevant if taken alone
Y : particularly relevant if combined with another document of the same category
A : technological background
O : non-written disclosure
P : intermediate document

T : theory or principle underlying the invention
E : earlier patent document, but published on, or after the filing date
D : document cited in the application
L : document cited for other reasons

& : member of the same patent family, corresponding document

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